WATERJET PUMP PERFORMANCE DETERMINATIONS

Arthur F. Garcia, Jr.
Mechanical Engineer
Propulsion Systems Analysis Branch
Naval Ship Engineering Center

The views expressed herein are the personal opinions of the author and are not necessarily the official views of the Department of Defense or of a Military Department.

Abstract

The following calculation procedure originated from the necessity of determining waterjet pump performance requirements for waterjet propelled sea vessels. The waterjet pump sizing procedure is basically in two parts. The purpose of the first part is to determine the requirements of flow rate and head required of any pump to produce the necessary thrust to achieve design speed. Secondly, where the pump requirements are known, studies to determine pump rpm, cavitation limitation, impeller diameter, case volume and wet weight can be made to estimate some detailed aspects of various pump types which might be applied. A computer program has been developed from this procedure which streamlines studying variations in design. The procedures and the computer program can also be used in evaluating waterjet systems presented in studies and proposals.

Notation

Symbol	Units	Definition
A	ft^2	Area
De	ft	Elevation deviation
Dimp	ft	Impeller diameter
E		Efficiency
fi		Inlet velocity ratio
g	ft/sec^2	Gravitational constant
gpm	gal/min	Flow rate per pump stage
HD	ft	Head
HL	ft	Head loss
HLd	ft	Differential head loss
NPSH	ft	Net positive suction head
Ns		Pump specific speed
Patm	ft	Atmospheric pressure
Pvap	ft	Vaporization pressure of water
Q	ft^3/sec	Flow rate
R		Jet Velocity Ratio
Rey		Reynolds number
RPM		Revolutions per minute
Subm	ft	Submergence
SUCT		Suction specific speed
T	lbs	Thrust
Uz	ft/sec	Impeller tip speed
V	ft/sec, kts.	Velocity
WHP	HP	Ideal horsepower (water horsepower)
$\beta 2$		Impeller blade discharge angle
$\gamma = 64$	lbs/ft^3	Density
ϕ		Flow coefficient
π		3.14159
$\rho = \gamma/g$		Specific density
σ		Cavitation factor
ψ		Si

Subscripts

Symbol	Units	Definition
av		Available at the eye of the pump
ex		At the pump exit
fs		Free stream
i		Inlet
int		Intake
j		Jet
k		Craft
nz		Nozzle
pmp		Pump
s		Stage
t		Total
tr		Transmission

TABLE OF CONTENTS

The Basic Approach

The flow rate and the head differential required of the waterjet pump are based on three inter-related equations. The flow rate is a function of the velocity of flow across the inlet area (equ. 1). The thrust required is dependent on the flow rate, the jet velocity and craft velocity (equ. 2). The head differential the pump must provide is a function of the head of the jet stream, the head of the free stream, and system losses (equ. 6).

The head differential the pump must produce is simplified by equation 3 as the absolute head at the pump exit (equ. 4) minus the absolute head available at the eye of the pump (equ. 5). Head of the jet and head of the free stream are defined by equations 7 and 8.

Inlet Velocity Ratio

The inlet velocity ratio is the ratio of the inlet velocity Vi to the free stream velocity Vk at the point the inlet area is measured (equ. 9). Inlet velocity is calculated from the inlet velocity ratio at design speeds. Flow rate is calculated from the inlet velocity at the location along the inlet where the inlet area is measured (equ. 10). Inlet velocity ratio is usually smaller than one at cruising speeds but can be greater than one at take-off or when accelerating.

Intake and Nozzle Losses

Intake losses include all friction and elevation effects by the inlet and ducting system which reduces the free stream head to the head available at the pump. Intake efficiency is essentially the ratio of the absolute head available at the pump to the free stream head in absolute pressure (equ. 11). From equation 5 and intake efficiency, an expression defining head loss of the intake can be written (equ. 12). The intake efficiency is a unique physical characteristic of each inlet and ducting system. Figures 4 and 5 show intake efficiency characteristics for two typical inlet-ducting systems.

Figure 5 shows intake efficiency versus craft speed for various inlet velocity ratios. Speeds at which cavitation occurs in the intake system can be indicated on the intake performance curves. The example curves tend to suggest that inlet velocity ratios greater than one have high intake efficiencies. However, intake efficiency will reflect head loss due to friction and elevation effects only. When the inlet velocity is greater than the free stream velocity the head at the inlet has to be greater than the free stream head. That head differential will be a head loss to the pump. So, when the inlet velocity is greater than the free stream velocity, an extra head loss factor for the inlet differential must be included in the head in determining pump head (equ. 13).

The nozzle efficiency term, Enz, expresses the ratio of the head of the jet absolute to the head at the exit of the pump where the nozzle is at a constant elevation only (equ. 14). Nozzle loss calculated from equation 16 will reflect frictional loss in the nozzle only. Equation 16 was developed from equation 4 and nozzle efficiency (equ. 15). Normally nozzles are short to minimize losses and do not significantly change in elevation with respect to the eye of the pump. The eye of the pump coincides with the maximun elevation of the ducting system. Since all intake losses are accounted for at that maximum duct elevation point, any further elevation losses must be measured with respect to that datum point. As shown in figure 8, elevation deviation of the nozzle can be measured from the eye of the pump. This method of compensating for elevation deviation does not reflect the internal frictional losses exactly since the elevation deviation will cause internal flow in piping to be faster or slower than at constant elevation. But elevation deviation will normally be very small with respect to jet head anyway, so the discrepancy will be insignificant. Nozzle efficiencies of .995 are typical for short nozzles (nozzle length is no longer than twice the throat diameter) where Reynolds number is freater than 400,000.

The basic pump head equation (equ. 6) can be modified to accept the above special losses.

$$HD_{pmp} = HD_j - HD_{fs} + HL_{nz} + HL_{int} + HL_d + D_e \quad (17)$$

Total Thrust Required

The total thrust required should be equal to the total drag effect at design craft speed. The total drag of the craft will include external drag effects of the strut and pod of a ram type inlet or the drag of a semiflush lip. An inlet area on a planing surface may affect planing characteristics which may affect trim and drag. The internal drag of the ducting is compensated for in the intake loss calculations. Where a study may include various inlet and duct sizes and shapes, the effect on drag should be inlcuded in determining the overall thrust required at design speed (figure 9). A check should be made on displacement and drag effect of reaction forces due to turns in ducting systems (figure 10). The drag or displacement effect of ducting turns may be significant in some craft at certain speeds. Also, where thrust is vectored, the total thrust required will be the same as the reaction resultant necessary to meet the lift and drag components.

Pump RPM and Cavitation Determinations

Figure 12 shows how head per stage and flow rate per stage are determined for the various pump types. Centrifugal pumps can be banked with several stages to give a lower flow rate per stage for a given pump head. Mixed-flow and axial pumps have the advantage of decreasing the head per stage at a given flow rate per pump. The pump performance curves of figure 13 were taken from pump technology (ref. 1, page 34) and indicate performance per stage. The performance curves indicate specific speeds at which the three types of pumps perform the best. Where a centrifugal pump is being considered for design at sustained conditions, the specific speed per stage should be between 500 and 2,000. Mixed-flow pumps should fall in specific speeds of 2,000 to 10,000. Axial pumps should operate at specific speeds of 10,000 to 15,000. If the pump type and geometry is fixed, the specific speed can be estimated from the appropriate pump geometry region. Knowing the flow per stage and the head per stage, the estimated pump rpm can be calculated from the estimated specific speed (equ. 19).

The net positive specific speed available at the eye of the pump is the absolute head available at the eye of the pump from previous calculations (equ. 5) minus the vaporization pressure of water (equ. 20). The cavitation factor sigma (σ) is calculated from the net positive suction head and the head per stage of the pump (equ. 21) (see ref 1, page 35). With centrifugal pumps, sigma applies for each stage since each stage must produce the total pump head. However, with axial and mixed-flow pumps, only the first stage "sees" the NPSH so sigma applies to the first stage only. Using NPSH and the rpm estimated from pump specific speed the estimated suction specific speed per stage can be calculated (equ. 22). As can be seen from figure 14, where SUCT is less than 8,000, the pump will be in a safe design region. Where SUCT is between 8,000 and 12,000, the pump will be a special design. Pumps having a suction specific speed between 12,000 and 20,000 will be in a critical design region. No conventional pumps are designed with suction specific speeds greater than 20,000. SUCT will be an indication of the complexity of pump design involved and whether a conventional design is possible with the given configuration.

Impeller Diameter Estimation

Impeller diameter is estimated by generalizing conventional pump design for the various pump types. A typical blade discharge angle for the three types can be assumed to be between $\beta 2$ = 20 degrees and $\beta 2$ = 22.5 degrees (see ref 1, pages 49 and 50). Figure 15 shows non-dimensional performance curves used in pump design (see ref. 1, page 61).

Pump performance curves can be simplified for the two typical blade discharge angles and plotted as the head coefficient versus the pump specific speed (figure 16). The head coefficient si (ψ) is a function of the head per stage and the impeller tip speed (U_2) in feet per second (equ. 23). By functionalizing si, impeller diameter can be calculated calculated from the impeller tip speed and the estimated impeller rpm (equ. 24). Equations exist where case diameter and case length are estimated from impeller diameter (reference 1, page 60). The casing volume is calculated from the casing dimensions and using a weight density factor for pump wet weight, the pump wet weight can be estimated (see Program Pring-out Appendix A.).

Computer Program

A computer program has been written which is based on the relationships developed in this paper. The computer program is written to allow entering a range of input so that output will show a trend when studying variations. The program is a basic tool; ie, a mechanism which only needs accurate imperical intake performance data and thrust requirement data to give accurate head, flow and horsepower values. The computer program also allows for estimating more detailed pump characteristics for various types of pumps with various stages. RPM, sigma, suction specific speed and impeller diameter are rough cut estimates which can serve to find critical areas, limitations and generally, the type of pump which may be best suited for a particular craft at design speeds. Wet estimates for aluminum type pumps are also part of the output. Wet weight estimates are based on density factors which may be outdated, too conservative or over estimated since some computational comparisons have shown that the computed wet pump weight has been twice what an actual on the shelf pump might weigh. The computer program included in this paper has the density factors which have shown this error. As more realistic wet weight factors become available, or a better method of estimating the wet weight of the pump becomes available, that part of the computer program should be revised and updated. Refer to Appendix A for a sample run, a program print-out and explanation of terms. The values calculated by the program have been validated by hand calculations.

Conclusion

The relationships defined in this paper were developed to support the computer program which has been developed for analyzing waterjet system performance. Using this computer program, the inlet area sizing diagram and the sensitivity chart in Appendix B and intake performance characteristics as shown in figures 4 and 5, a detailed study can be conducted which could determine the most optimum system for a given design speed. Once the inlet area, ducting configuration and pump type are fixed the program can be used to check for performance characteristics and limitations at hump speed, full speed and other important craft speeds. The procedure and the program are still in the development stage and are being submitted as the basic pattern to be improved as design data becomes available.

ACKNOWLEDGEMENTS

Berg, David J.
Brandau, John H.
Corso, James
Hale, Malcom
Johnson, Anthony
Lombardi, Paul
Jones, Walt
Kimble Nancy (typist)
Williamson, Carol (typist)

REFERENCES

1) "Waterjet Propulsion System Study, Pump Selection and Design", Lockheed of California Report LR 17885-2 (Report No. 2). From contract NObs 88605, June 1963

2) "Waterjet Propulsion System Study, Internal Flow Test", Lockheed of California Report LR 17885-3 (Report No. 3) From contract NObs 88605, June 1963

3) "Waterjet Propulsion System Study, System Design and Analysis", Lockheed of California Report LR 17885-5 (Report No. 5). From contract NObs 88605, June 1963

4) "Waterjet Propulsion - A critical Survey of the State of the Art and a Recommended Research and development Program". Technical Note 70, Hydromechanics Laboratory, Naval Ship Research and Development Center; April 1967

5) "Fluid Mechanics," third edition, by R.C. Binder, Prentice-Hall, N.J. 1960

APPENDIX A

Sample Run

The computer program is presently on a time sharing system. As indicated by the sample run figures A1 and A2, on-line input happens to be one of the features of the system used. The question marks indicate readyness for input according to headings as programmed. Pump efficiency and transmission efficiency are inputs which are used to calculate a pump horsepower and a shaft horsepower. An estimated pump efficiency and transmission efficiency is entered when the exact value is not known. The program is arranged to calculate for any combination of inlets, pumps and jets the craft may use. Most of the headings are self explanatory however, the following terms might be clarified: VEL RATIO is jet velocity ratio, MIL RE/NZL is Reynolds number at the nozzle throat in millions, AVLHD FT is the gage pressure available at the eye of the pump in feet. IDL EF is ideal system efficiency, IMPLR GAL/M is flow rate per stage in gallons per minute, WET WT is pump wet weight for an aluminum type pump in thousands of pounds.

List of Terms in Computer Program

ARINL	Ai
ARJET	Aj
AVLHD	HD avg
DC	Pump case dia
DIAM	Dimp
D2, Dj^2	Nozzle throat dia squared
D22,	Mean impeller dia squared
ED	De
EFIDL	Ideal system efficiency
EFINT	Eint
EFNZL	Enz
EFPMP	Epmp
EFTRN	Etr
EHP	Effective HP
FLO1, Q1	Flow rate ft^3/min
FLO2, Q2	Flow rate lb/sec
FLO3, Q3	Flow rate gal/min
FLO4	gal/min per stage
FSHD	HDfs
HDPMP	HDpmp
HLD	HLd
HLINT	HLint
HLNZL	HLnz
ICNTL	control
MODE	control
NTYPE	control
OPENS, Cpns	Number of inlets
PATM	Patm
PC	Propulsion Coefficient
PHP	Pump HP

PMPHD	HDpmp
PMPS, Pmp(s)	Number of pumps
PNPSH	NPSH
REJET, Rey	Reynolds number
RPM	Revolutions per min
SHP	Shaft HP per pump
SHDT, SHPt	Total SHP
SI	
SIG	
STGS	Number of stages per pump
SUBM	Subm
SUCT	Suction specific speed
TR, Tr	Tt
U2	Impeller tip speed
VI	Vi
VIOVK	fi
VJ	Vj
VJOVK	R
VK	Vk(Kts)
VOL	Pump case volume
WC	Case wet density
WHP2	WHP
WWAL	Wet weight of aluminum type pump
XIST, Tst	Static thrust
XLC	Pump case length
XNS	Ns
Z	Axial pump length factor

```
IN .FIRST

  VK(KTS)   INLETS   PUMPS   JETS   SUBM(FT)
?  45.        2        2       2      2.

  PUMP EFF   TRNSMSN EFF   NOZZLE EFF   ELEV DEV(FT)
?  .87          1.            .995          0.0

   MODE=1- ADDITIONAL ENTRY , =0- LAST ENTRY

  AREA/INL(FT2)   VI0VK   INTK EFF   TOT THRUST REQ'D(LBS)   MODE
?   .455           .7       .45          10628.                1
?   .54            .7       .45          10800.                0

   A/INL     VI      VEL       VJ       JET     A/NZL   MIL RE/
    FT2     FT/S    RATIO     FT/S      EFF      FT2      NZL
    .45    53.20    2.45    186.29    .5795     .13      5.87
    .54    53.20    2.24    170.43    .6168     .17      6.12

  PER PUMP VALUES

     A/INL      FLO RT     FLO RT     FLO RT     IDEAL STAT
      FT2        LBS/S      FT3/M      GAL/M      THRUST-LB
      .45       1550.25    1452.40   10863.98      8976.01
      .54       1839.85    1723.73   12893.52      9746.12

 FREE STREAM HEAD (FT-CAGE)=      91.7674

   A/INL     NZL      INTK     AVL HD    PMP HD    NPSH      WHP
    FT2    LOS-FT    LOS-FT      FT        FT       FT        HP
    .45     2.88     69.12     22.64    519.55    54.75   1464.41
    .54     2.44     69.12     22.64    431.20    54.75   1442.46

 A/INL-FT2     EHP     PHP/PMP    SHP TOTL      PC      IDL EFF
    .45     1468.64   1683.23    3366.46     .4363     .5014
    .54     1492.41   1658.00    3316.01     .4501     .5173

 ENTER CONTROL=1 FOR RPM AND DIA ESTIMATES, CONTROL= 2 TO SKIP
  HEADING, CONTROL= 0  TO STOP PROGRAM.   CONTROL
?   1
```

```
DESIGN ESTIMATES BASED ON CONVENTIONAL PUMP PERFORMANCE

  CENTRIFUGAL MULTISTAGE PUMPS (TYPE 1)
    ASSUMPTIONS: BETA=22.5 DEG,CASE DIA=1.90X IMPLR DIA,
    CASE VOL BASED ON DOUBLE SUCTION IMPLRS

  MIXED-FLOW PUMPS (TYPE 2)
    ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.50X IMPLR DIA
    ONE STAGE ONLY

  AXIAL-FLOW PUMPS (TYPE 3)
    ASSUMPTIONS: BETA=20 DEG, CASE DIA=1.20X  IMPELLER DIA
    6 BLADES/ STAGE

  PUMP WET WT WILL BE 167 PER CENT OF THE ALUMINUM WET WT
FOR STAINLESS STEEL OR OTHER BASIC PUMP MATERIALS

 VOL AND WT ARE BASED ON THE ESTIMATED DIAMETER
  KLB IS THOUSANDS OF LBS
     NS=0  STOPS PROGRAM

  TYPE      NS        IMPLRS OR STAGES / PUMP
?   1      1400.            2

A/INL  SIG     IMPLR      RPM      SUCT     IMPLR     CASE     WET WT
 FT2           GAL/M               SP SPD   DIA-FT    VOL-FT3  ALM-KLB
 .45 .1054    5431.99   2067.13   7568.85    1.68     29.34     2.79
 .54 .1270    6446.76   1649.95   6581.49    1.92     43.62     4.15

  TYPE      NS        IMPLRS OR STAGES / PUMP
?  0        0          0
```

Calculation Procedure

Input:

V_k(Kts), O_{pns}, P_{mps}, I_{ts},

S_{ubm}, EF_{noz}, D_e, A_i (area per inlet)

f_i, E_{int}, T_r,

Calculations:

1) $V_i = (V_i/V_k)(V_k \times 1.688)$

$V_i = f_i V_k \times 1.688$, inlet velocity

2) $T_r = \rho Q (V_j - V_k)$

$Q = AV = A_i V_i = A_{nz} V_{nz}$

$T_r = \rho$ (total inlet area) $V_i (V_j - V_k)$

$T_r = \rho O_{pns} \times A_i V_i (V_j - V_k) \left(\frac{V_k}{V_k}\right)$

$V_j/V_k - 1 = T_r/(\rho O_{pns} A_i V_i V_k \times 1.688)$

$V_j/V_k = 1 + T_r/(\rho O_{pns} A_i V_i V_k \times 1.688)$

jet velocity ratio

3) $V_j = (V_j/V_k)(V_k \times 1.688)$, jet velocity

4) $E_j = \dfrac{2}{1 + V_j/V_k}$, jet efficiency

5) $A_j = Q/V_j = A_i O_{pns} V_i/(V_j \times Jts)$

$A_j = A_i O_{pns}/(Jts \times V_j/V_k)$,

throat areaper nozzle

6) $A_j = \pi D_j^2/4$

$D_j = 4A_j/\pi$, nozzle dia

$Rey = \dfrac{V_j D_j}{\nu}$, reynolds no.

$\nu = 12.9 \times 10^{-6}$ ft^2/sec @ 59°F sea water

7) $Q_1 = 60.\ A_i\ O_{pns}\ V_i/P_{mps}$, ($ft^3$/min)

flow rate per pump

$Q_2 = Q_1\ \gamma/60.$, (lbs/sec)

$Q_3 = Q_2\ (7.48)$, (gal/sec)

8) $T_{st} = Q\ (V_j - 0)$, ideal static thrust

9) $HD_{fs} = (V_k \times 1.688)^2/2g + S_{ubm}$

free stream HD

10) $HL_{nz} = (1-E_{nz})\ (V_j^2/2g + P_{atm})/E_{nz}$

$HL_{ns}/P_{mp} = HL_{nz}\ J_{ts}/P_{mps}$

nozzle head loss per pump

11) $HL_{int} = (1-E_{int})\ (HD_{fs} + P_{atm})$

$HL_{int}/P_{mp} = HL_{int} \times O_{pns}/P_{mps}$

intake head loss per pump

12) When $V_i > V_k \times 1.688$:

$HL_d = (V_i^2/2g - (V_k \times 1.688)^2/2g$

$HL_d/P_{mp} = HL_d \times O_{pns}/P_{mps}$

inlet differential head loss per pump

13) $HD_{avg}/P_{mp} = HD_{fs} - HL_{int}/P_{mp} - HL_d/P_{mps}$

head available (gage) per pump

14) $D_e/P_{mp} = D_e J_{ts}/P_{mps}$

elevation deviation head per pump

15) $HD_{pmp} = V_j^2/2g + HL_{nz}/P_{mp} - HD_{avg}/P_{mp}$

$+ D_e/P_{mp}$

head differential each pump will have to provide

16) NPSH = $HD_{avg} + P_{atm} - P_{vap}$

$HD_{avg} + 32.11$

net positive suction head

17) WHP = $Q_1 \times HD_{pmp}/550$, water - horsepower or the ideal horsepower per pump

18) EHP = $T_r \ (V_k \times 1.688)/550$ effective horsepower or resistance horsepower

19) E_{idl} = $EHP/(P_{mps} \times WHP)$, ideal system efficiency

20) PHP = WHP/E_{pmp}, pump horsepower

21) SHP = PHP/E_{tr} shaft horsepower per pump

22) SHP_t = $SHP \times P_{mps}$, total SHP

23) PC = EHP/SHP_t

At this point, the type of pump, the pump specific speed and the number of stages per pump must be entered. Head and flow rate per stage is then calculated according to the pump type as shown in figure 12. RPM is then calculated from equation 19. NPSH is retrieved from previous calculations and used to calculate sigma and SUCT as shown in equations 21 and 22. Then using a function for si versus pump specific speed, tip speed can be calculated (U_2) and used in calculating impeller diameter as illustrated in figures 15 and 16. From the impeller diameter, casing diameter and casing length are calculated according to size estimation factors given in reference 1 (page 60). Refer to the computer program print-out for the size relationships used. The computer program is simple to read and is straight-forward in arrangement.

```
WAIT.

02C     WJPA (WATER-JET PUMP PART A) CASDAC NO.-
03C     PREDICTS WATER JET PUMP REQUIREMENTS FOR A GIVEN APPLICATION
04C     AND SHOULD BE USED WITH WJPB FOR RPM AND DIA ESTIMATES
10     DIMENSION  ARINL(20), VI0VK(20), EFINT(20), TR(20)
20     DIMENSION  VI(20), VJ(20), ARJET(20), FLO2(20)
30     DIMENSION   FLO3(30), PNPSH(20), HDPMP(20), WHP2(20)
35     DIMENSION   EFJET(20)
40     CONV=1.68894
45     PI=3.14159
50     VISC=12.9
55     DENS=64.042
57     PATM=33.91
60     G=32.174
65     RHO=1.9905
70     PRINT:::"   VK(KTS)   INLETS   PUMPS   JETS   SUBM(FT)"
90     INPUT, VK, OPENS, PMPS, XJTS, SUBM
100     PRINT:"   PUMP EFF   TRNSMSN EFF   NOZZLE EFF   ELEV DEV(FT)"
110     INPUT, EFPMP, EFTRN, EFNZL, ED
120     I=1
130     PRINT:"     MODE=1- ADDITIONAL ENTRY , =0- LAST ENTRY"
140     PRINT:"   AREA/INL(FT2)   VI0VK   INTK EFF   TOT THRUST
150     + REQ'D(LBS)   MODE"
160 10 INPUT, ARINL(I), VI0VK(I), EFINT(I), TR(I), MODE
165     N=I
170     I=I+1
180     IF(MODE) 20, 20, 10
190 20 PRINT:"      A/INL       VI        VEL         VJ          JET
195     +     A/NZL    MIL RE/"
210     PRINT "          FT2        FT/S       RATIO        FT/S        EFF
215     +       FT2          NZL"
220     DO 25  J=1,N
230     VI(J)=VI0VK(J)*VK*CONV
235     VJ0VK=1.+TR(J)/(RHO*ARINL(J)*OPENS*VI(J)*VK*CONV)
240     VJ(J)=VJ0VK*VK*CONV
245     EFJET(J)=2./(1.+VJ0VK)
250     ARJET(J)=ARINL(J)*OPENS/(VJ(J)*XJTS/VI(J))
255     D2=4.*ARJET(J)/PI
260     REJET=VJ(J)*SQRT(D2)/VISC
265 25 PRINT 3,ARINL(J),VI(J), VJ0VK,VJ(J),EFJET(J),ARJET(J),REJET
267      PRINT:"    PER PUMP VALUES"
270     PRINT:"          A/INL           FLO RT          FLO RT
275     +          FLO RT    IDEAL STAT"
280     PRINT "           FT2            LBS/S           FT3/M
285     +          GAL/M       THRUST-LB"
290     DO 30  J2=1, N
295     FLO1=60. *ARINL(J2)*OPENS*VI(J2)/PMPS
300     FLO2(J2)=FLO1*DENS/60.
305     FLO3(J2)=FLO1*7.48
310     XIST=FLO2(J2)*VJ(J2)/G
320 30 PRINT 4, ARINL(J2), FLO2(J2), FLO1, FLO3(J2), XIST
```

```
324     FSHD=(VK*CONV)**2/(2.*G)+SUBM
326     PRINT:"   FREE STREAM HEAD (FT-GAGE)=", FSHD
340     PRINT:"      A/INL        NZL       INTK     AVL HD    PMP HD
350     +       NPSH         WHP"
360     PRINT "          FT2     LOS-FT     LOS-FT          FT          FT
370     +        FT         HP"
380     DO 40  K=1,N
390     HLNZL=(1.-EFNZL)*((VJ(K)**2/(2.*G)+PATM)*XJTS)/(PMPS*EFNZL)
410     HLINT=(1.-EFINT(K))*(FSHD+PATM)*OPENS/PMPS
412     HLD=0.0
414     B=VI(K)**2
416     C=(VK*CONV)**2
418     IF(B-C) 37, 37, 35
420 35  HLD=(B-C)*OPENS/(2.*G*PMPS)
422 37  AVLHD=FSHD-HLINT-HLD
430     HDPMP(K)=VJ(K)**2/(2.*G)-AVLHD+HLNZL+ED*XJTS/PMPS
435     PNPSH(K)=AVLHD+32.11
450     WHP2(K)=HDPMP(K)*FLO2(K)/550.
460 40  PRINT 5, ARINL(K),HLNZL,HLINT,AVLHD,HDPMP(K),PNPSH(K),WHP2(K)
470     PRINT:"  A/INL-FT2          EHP     PHP/PMP    SHP TOTL
480     +       PC     IDL EFF"
490     DO 45  L=1,N
500     EHP=TR(L)*VK*CONV/550.
510     PHP=WHP2(L)/EFPMP
520     SHP=PHP/EFTRN
530     SHPT=SHP*PMPS
540     PC=EHP/SHPT
550     EFIDL=EHP/(WHP2(L)*PMPS)
560 45  PRINT 7, ARINL(L), EHP, PHP, SHPT, PC, EFIDL
570     PRINT:"  ENTER CONTROL=1 FOR RPM AND DIA ESTIMATES, CONTROL=
580     + 2 TO SKIP"
590     PRINT"    HEADING, CONTROL= 0  TO STOP PROGRAM.   CONTROL "
600     INPUT , ICNTL
610     IF(ICNTL-1) 47, 48, 50
615 48  CONTINUE
620     PRINT:" DESIGN ESTIMATES BASED ON CONVENTIONAL PUMP PERFORMANCE"
630     $USE WJPB
640 47  STOP
700     3 FORMAT(4F10.2, F10.4, 2F10.2)
710   4 FORMAT(5F14.2)
720   5 FORMAT(7F10.2)
730   7 FORMAT(4F11.2,2F11.4)
740   9 FORMAT(F5.2, F5.4, 6F10.2)
```

```
10C    WJPB (WATER-JET PUMP PART B) CASDAC NO.-
20C    MUST BE USED WITH WJPA TO ESTIMATE RPM AND DIA FOR WATER-
30C    JET PUMPS BASED ON CALCULATIONS FROM WJPA.
40C         A. F. GARCIA   NAVSEC 6144    10 JUNE 1969
50    PRINT:"    CENTRIFUGAL MULTISTAGE PUMPS (TYPE 1)"
60    PRINT "      ASSUMPTIONS: BETA=22.5 DEG,CASE DIA=1.90
70    +X IMPLR DIA,"
80    PRINT "      CASE VOL BASED ON DOUBLE SUCTION IMPLRS"
90    PRINT:"    MIXED-FLOW PUMPS (TYPE 2)"
100    PRINT "      ASSUMPTIONS: BETA=22.5 DEG, CASE DIA=1.50
110    +X IMPLR DIA"
120    PRINT "      ONE STAGE ONLY"
130    PRINT:"    AXIAL-FLOW PUMPS (TYPE 3)"
140    PRINT "      ASSUMPTIONS: BETA=20 DEG, CASE DIA=1.20X
150    + IMPELLER DIA"
160    PRINT "      6 BLADES/ STAGE"
170    PRINT:"   PUMP WET WT WILL BE 167 PER CENT OF THE ALUMINUM
180    + WET WT"
190    PRINT "FOR STAINLESS STEEL OR OTHER BASIC PUMP MATERIALS"
200    PRINT:"  VOL AND WT ARE BASED ON THE ESTIMATED DIAMETER"
205   PRINT "   KLB IS THOUSANDS OF LBS"
210    PRINT"      NS=0  STOPS PROGRAM"
```

```
220 50 PRINT:"   TYPE       NS        IMPLRS OR STAGES / PUMP"
230    INPUT, NTYPE, XNS, STGS
240    IF(XNS-500.)  95, 55, 55
250 55 PRINT:"A/INL  SIG      IMPLR        RPM       SUCT      IMPLR
260    +        CASE     WET WT"
270    PRINT "  FT2              GAL/M                SP SPD    DIA-FT
280    +   VOL-FT3   ALM-XLB"
290    DO 90  M=1,N
300    PMPHD=HDPMP(M)
305    FLO4=FLO3(M)/STGS
310    IF (NTYPE-2) 65, 65, 60
320 60 PMPHD=PMPHD/STGS
330    FLO4=FLO3(M)
340 65 SIG=PNPSH(M)/PMPHD
350    RPM=XNS*(PMPHD**.75)/FLO4**.5
360    SUCT=RPM*(FLO4**.5)/(PNPSH(M)**.75)
370    X=XNS/1000.
380    SI=.5883-X*7.1528E-2+(X**2)*3.8194E-3-(X**3)*6.944E-5
385    + +.01
390    U2=PMPHD*G/SI
400    DIAM=SQRT(U2)*60./(PI*RPM)
405    IF(NTYPE-2) 70, 72, 75
410 70 DC=1.5*DIAM
420    XLC=STGS*1.5*DIAM/2.+2.*DIAM
430    WC=.055
440    GO TO 85
460 72 DC=1.5*DIAM
470    XLC=2.7*DIAM
480    WC=.041
490    GO TO 85
500 75 D22=2.*DIAM**2/1.25
510    Z=.364*.80*PI*SQRT(D22)/6.
520    DC=1.2*SQRT(D22)
530    XLC=2.4*Z*STGS+2.2*SQRT(D22)
540    WC=.039
550 85 VOL=XLC*PI*DC**2/4.
560    WWAL=VOL*WC*1.728
570    PRINT 9, ARINL(M), SIG, FLO4, RPM, SUCT, DIAM, VOL, WWAL
580 90 CONTINUE
590    GO TO 50
600 95 CONTINUE
```

APPENDIX B

Inlet Area Sizing

A method of estimating realistic inlet area for certain craft is necessary since an infinite range of inlet areas and hence flow rates in combination with appropriate jet velocities could exist to meet the required thrust. From the thrust and flow equations (1) and (2):

$$T_r = \rho Q(V_j - V_k)$$

$$Q = A_i V_i, \quad V_i = f_i V_k$$

$$T_r = \rho f_i A_i V_k (V_j - V_k)$$

$$T_r = \rho f_i A_i V_k^2 (V_j/V_k - 1)$$

$$T_r / f_i A_i = \rho V_k^2 (R-1).$$

If:

$$\text{Total thrust}/f_i(\text{Total inlet area}) = \rho V_k^2 (R-1)$$

Then:

$$\text{Thrust per nozzle}/f_i(\text{Total inlet area per nozzle}) = \tau = \rho V_k^2 (R-1)$$

Tau (τ) is then a function of the inlet area required per jet nozzle on the craft to meet a particular thrust. Tau is also a function of jet velocity and jet velocity ratio. By generating a family of curves of tau versus ship speed and jet velocity ratio, inlet area can be determined from tau where the design speed and design jet velocity ratio are known and a certain thrust per nozzle is required (fig. B.1). Regions of typical design velocities and design jet velocity ratios for certain waterjet propelled craft can be shown on figure B.1. The conventional design regions can be used for "first-cut" estimations of inlet area from tau for specific craft. The design regions for the various craft as shown in figure B.1 are general and based on limited information and should be revised as more exact design jet velocity ratios are known at design speeds for the various craft.

Sensitivity Trends

The required pump head differential and flow rate are sensitive to the jet velocity ratio. It can be shown that for low jet velocity ratios ($R \rightarrow 1$) the ideal jet efficiency will be high, but the flow rate required will become very high and the pump head differential will become unrealistically low. However, as the jet velocity ratio increases, the ideal jet velocity ratio will drop so as $R \rightarrow 3$, $\epsilon_j = .50$ An ideal jet efficiency of fifty per cent may be undesireably low for design cruising speeds for most craft. For most applications then, R will be between one and three.

$$\epsilon_j = \frac{2}{1+R}$$

From equations (2) and (4):

$$T_r = \rho\ Q\ (V_j - V_k)$$

$$T_r = \rho\ AifiV_k^2\ (R-1)$$

If T_r and V_k are constant,

$$Ai \approx \frac{1}{fi\ (R-1)}$$

And,

$$Q = ViAi = fiV_kAi$$

so,

$$Q \approx \frac{fiVk}{fi(R-1)}$$

And since V_k is constant,

$$Q \approx \frac{1}{(R-1)}$$

The equation for pump head (equ. 6) can be simplified to be:

$$HDpump = Vj^2/2g + Eint\ (V_k^2/2g) + C$$

"C" is miscellaneous head loss such as nozzle submergence, and deviation loss which tends to be small and constant. If "C" can be considered insignificant with respect to the total head the pump must produce, let C = 0 then:

$$HDpmp = \frac{1}{2g} (V_j^2 + Eint\ V_k^2)$$

$$= \frac{V_k^2}{2g} (V_j^2/V_k^2 + Eint)$$

$$HDpmp \approx R^2 = Eint$$

Now,

$$WHP = Q \times HDpmp$$

From the specific speed equation for pumps, the RPM can be estimated (equ. 19):

$$RPM = \frac{Ns\ HDpmp^{.75}}{\sqrt{gpm}}$$

But gpm is just flow rate Q, so if Ns is constant:

$$RPM \approx \frac{HDpmp^{.75}}{\sqrt{Q}}$$

The above equation shows the effect of head and flow rate on pump RPM where pump specific speed is kept constant (pump geometry, or the pump type is kept constant). Suction specific speed can then be related to the RPM (equ. 22):

$$SUCT = \frac{RPM\ \sqrt{gpm}}{(NPSH)^{.75}}$$

or:

$$SUCT = \frac{Ns\ HDpmp^{.75}\ \sqrt{gpm}}{\sqrt{gpm}\ (NPSH)^{.75}}$$

Then if NPSH is constant:

$$SUCT \approx HDpmp^{.75}$$

Also from equation 21:

$$\sigma = \frac{NPSH}{HDpmp}$$

so:

$$\sigma \approx \frac{1}{HDpmp}$$

The above relationships can be depicted graphically as shown in figure B.2 as relative change in magnitude versus change in jet velocity ratio. Trends in head, flow, ideal horsepower, pump RPM, and suction specific can be seen as the design jet velocity ratio changes. The thrust required, pump specific speed and net positive suction speed must be held constant which means that duct configuration and the pump type (pump geometry) is kept constant. See the following Sensitivity Table as an example of the effect of changing a system from a design velocity ratio of 2 to 3.

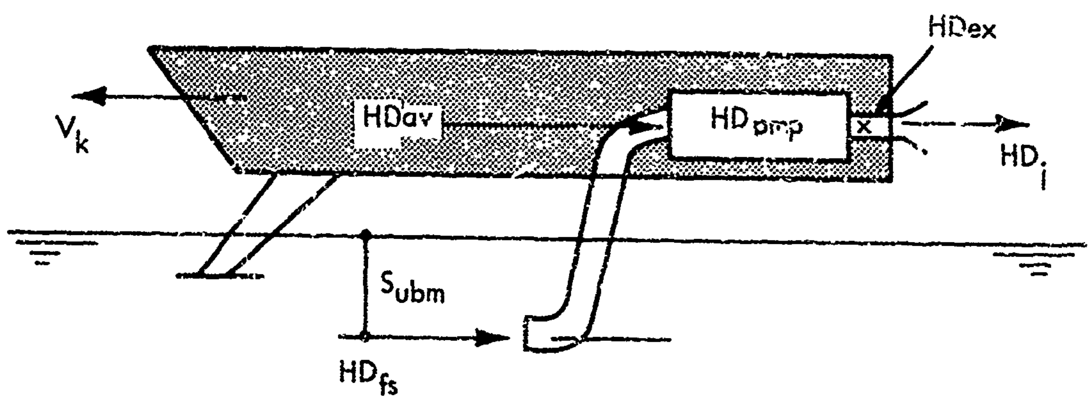
Sensitivity Table

	R = 2	R = 3	% change from R = 2
A_i, f_i = 1.0	1	.5	50%
Q	1	.5	50
HDpmp, Eint = 1.0	3	8	266
WHP	3	4	133
RPM	2.3	6.1	265
SUCT	2.3	4.8	207
σ	.5	.19	.38

<u>BASIC WATERJET EQUATIONS</u>

$$Q = A_j V_j \quad (1)$$

$$T_t = \rho Q (V_j - V_k) \quad (2)$$



$$HDpmp = HDex - HDav \quad (3)$$

$$HDex = HDj + HLnz + Patm \quad (4)$$

$$HDav = HDfs - HLint + Patm \quad (5)$$

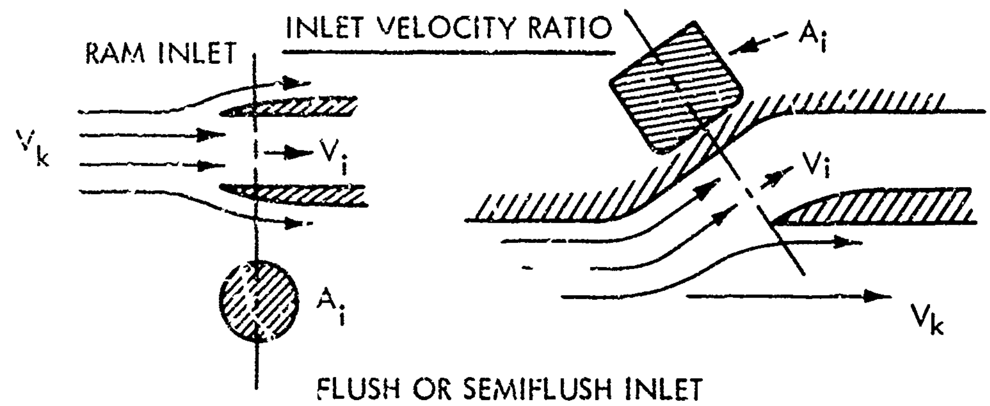
$$HDpmp = HDj + HLnz + Patm - (HDfs - HLint + Patm)$$

$$HDpmp = HDj - HDfs + HLnz + HLint \quad (6)$$

$$HD_j = V_j^2 / 2g \quad (7)$$

$$HDfs = V_k^2/2g + Subm \quad (8)$$

FIG. 1

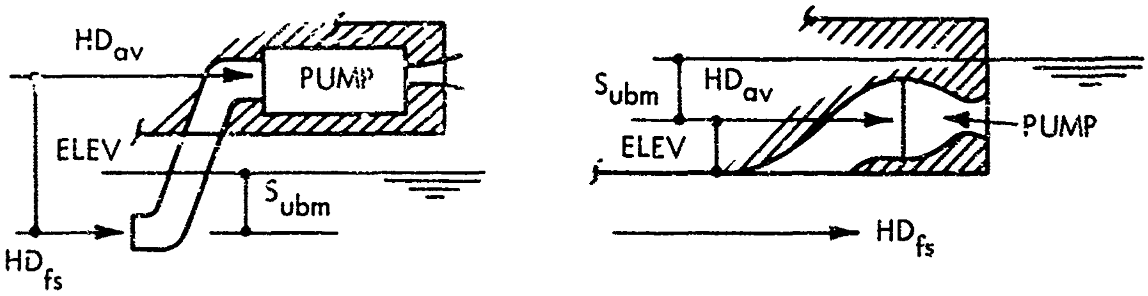


$$f_i = V_i / V_k \tag{9}$$

$$Q = A_i V_k f_i \tag{10}$$

FIG. 2

INTAKE LOSS



INTAKE EFFICIENCY:

$$E_{int} = \frac{HDav}{Vk^2/2g + S_{ubm} + Patm} = \frac{HDav}{HDfs + Patm} \tag{11}$$

$$HDav = Eint\left(HDfs + Patm\right)$$

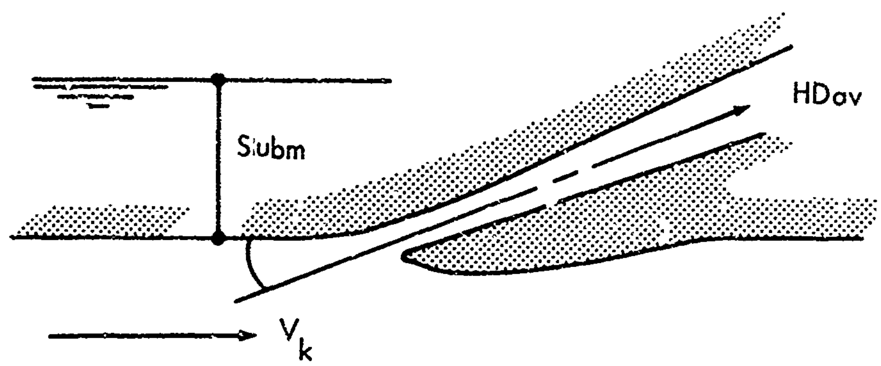
FROM EQU. (5)

$$HL\,int = HDfs + Patm - HDav$$

$$HLint = HDfs + Patm - Eint\left(HDfs + Patm\right)$$

$$HLint = \left(1 - Eint\right)\left(HDfs + Patm\right) \tag{12}$$

FIG. 3



FLUSH & SEMIFLUSH PERFORMANCE CURVES

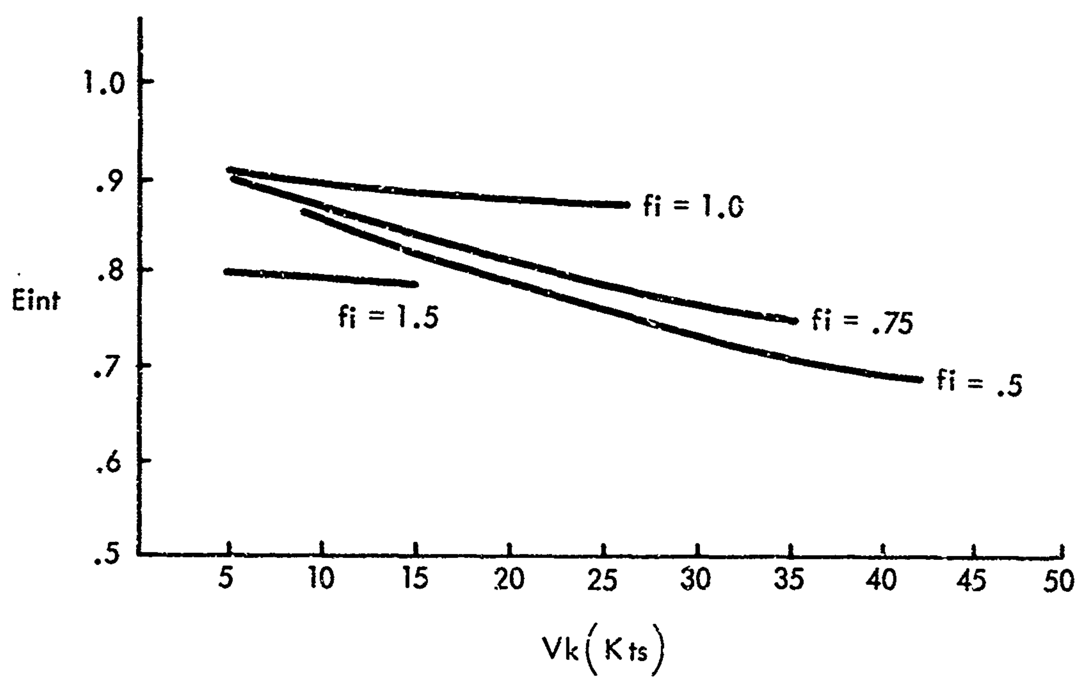


FIG. 4

STRUT INTAKE PERFORMANCE CURVES

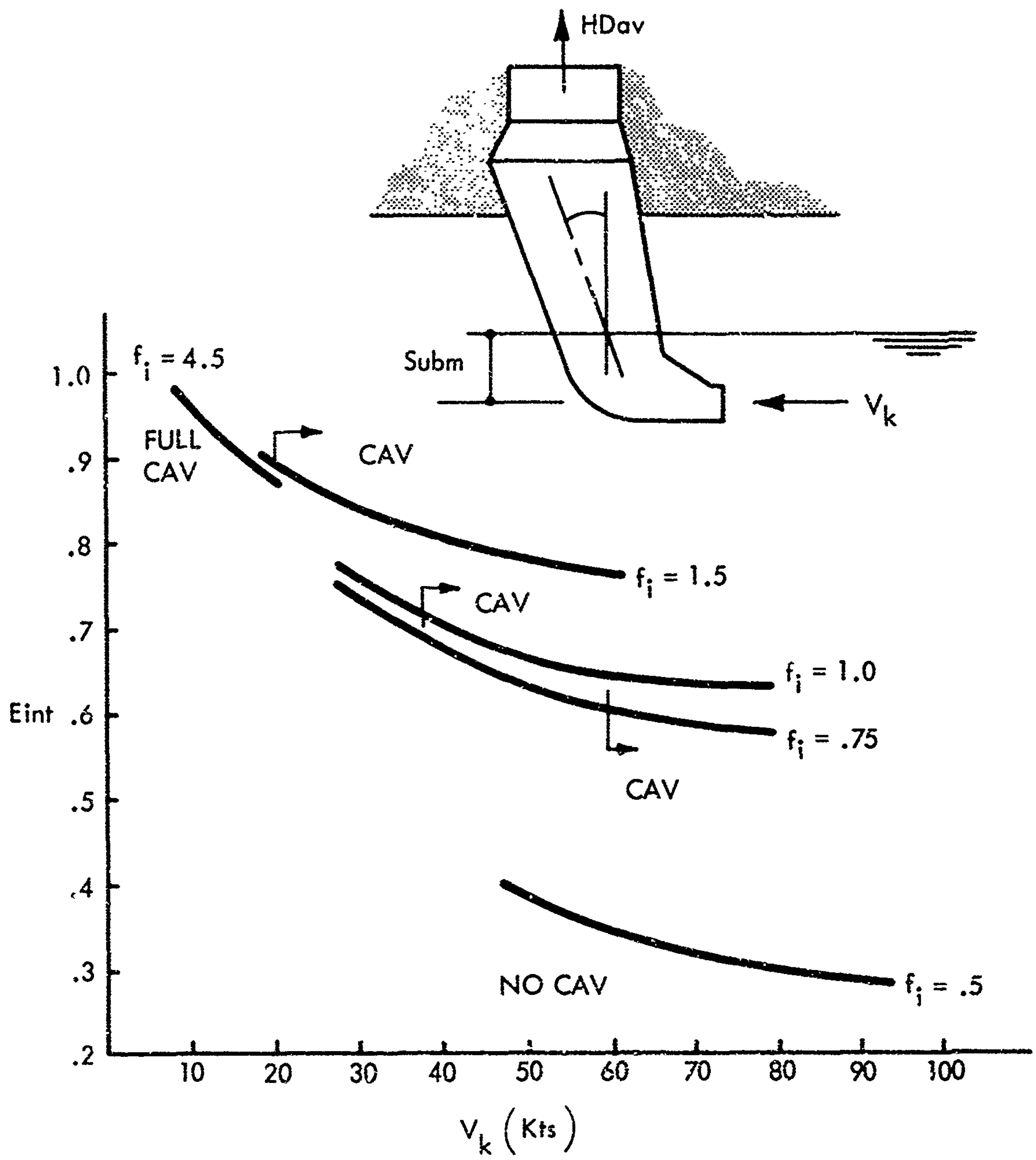
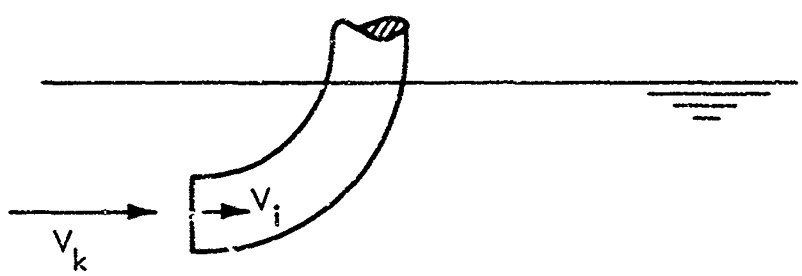


FIG. 5

HEAD DIFFERENTIAL LOSS AT INLET



WHEN $V_i > V_k$:

$$V_i^2/2g > V_k^2/2g$$

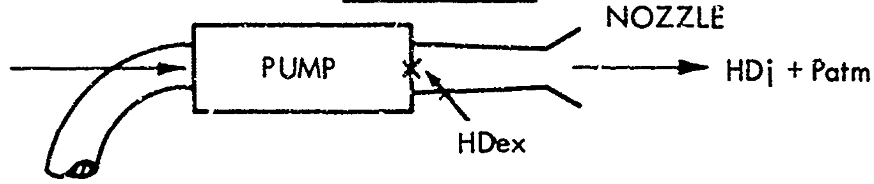
$$HL_d = \frac{V_i^2}{2g} - \frac{V_k^2}{2g}$$

$$HL_d = \frac{V_k^2}{2g}\left(\frac{V_i^2}{V_k^2} - 1\right)$$

$$HL_d = V_k^2\left(f_i^2 - 1\right)/2g \qquad (13)$$

FIG. 6

NOZZLE LOSS



NOZZLE EFFICIENCY :

$$Enz = \frac{HD_j + Parm}{HDex} \tag{14}$$

$$HDex = \frac{HD_j + Patm}{Enz} \tag{15}$$

FROM EQU. (4)

$$HLnz = HDex - \left(HD_j + Patm\right)$$

$$HLnz = \frac{HD_j + Patm}{Enz} - \left(HD_j + Patm\right)$$

$$HLnz = \left(HD_j + Patm\right)\left(1 - Enz\right) / Enz \tag{16}$$

FIG. 7

ELEVATION DEVIATION OF THE NOZZLE

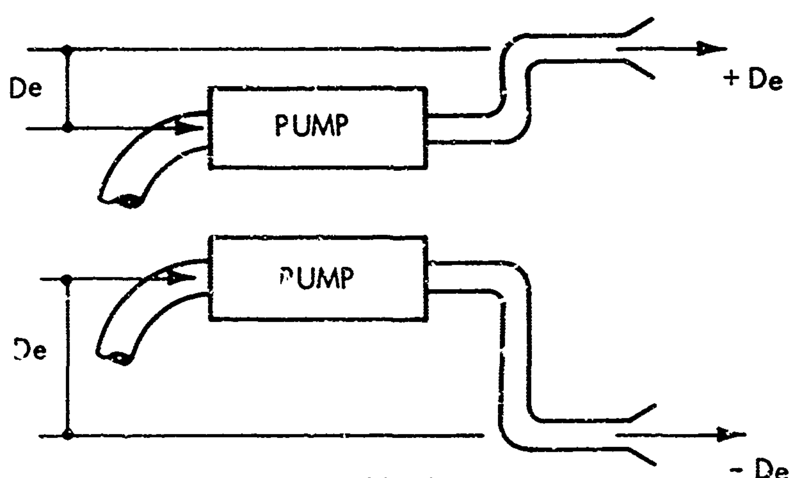
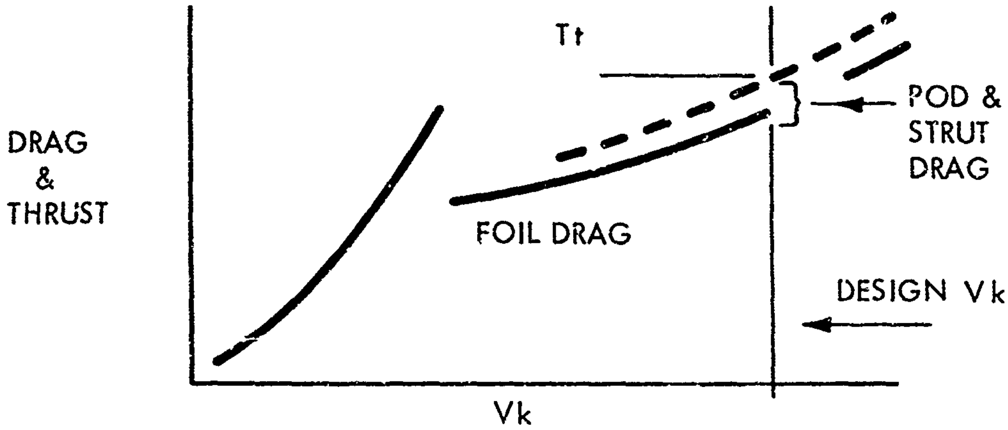
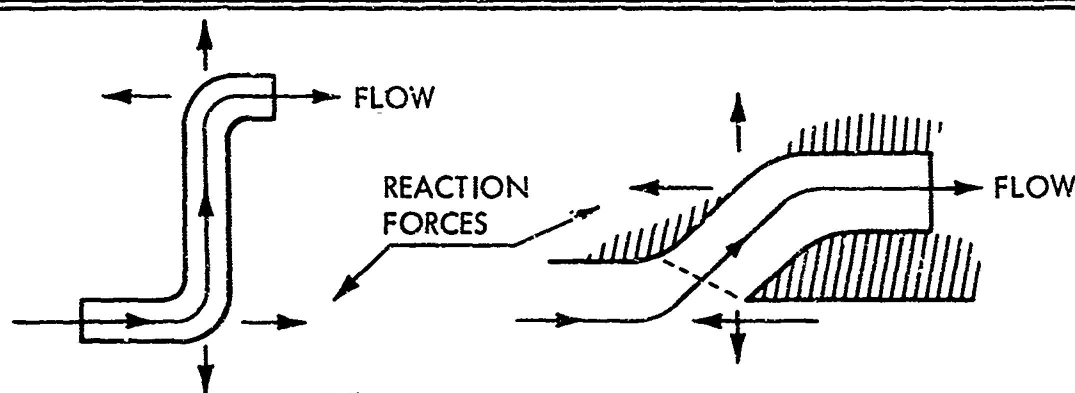


FIG. 8



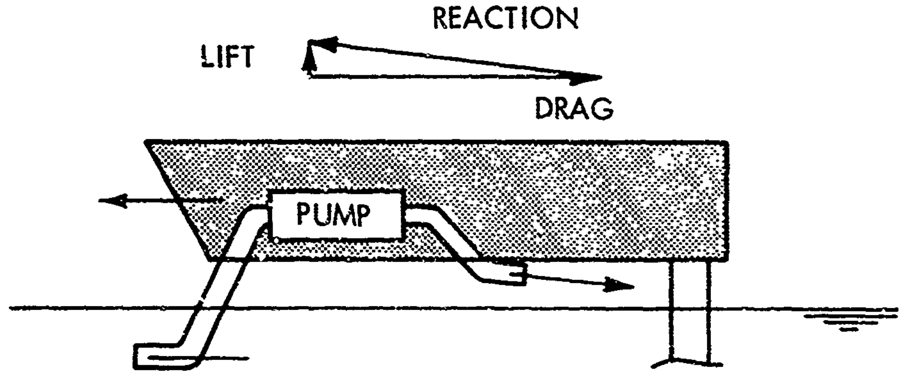
T_t = TOTAL DRAG AT DESIGN Vk

FIG. 9



INLET - DUCT REACTION FORCES

FIG. 10



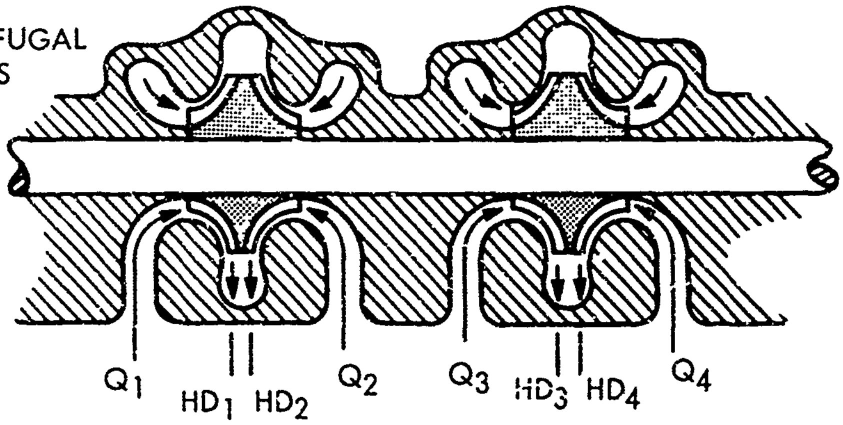
Tt = REACTION = $\sqrt{LIFT^2 + DRAG^2}$

VECTORED THRUST

FIG. 11

HEAD AND FLOW RATE PER STAGE

I. CENTRIFUGAL PUMPS

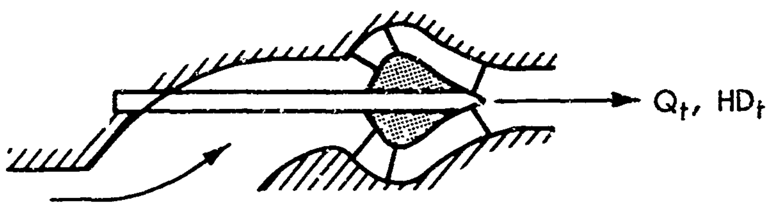


A. $Q_t = Q_1 + Q_2 + Q_3 + Q_4$

B. $Q_s = Q_t / Stgs$

C. $HD_s = HD_1 = HD_2 = HD_3 = HD_4 = HD_t$

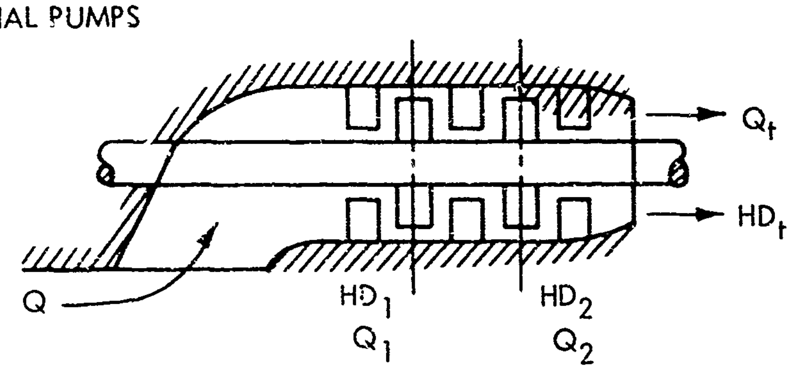
II. SINGLE-STAGE MIXED FLOW PUMPS



A. $Q_s = Q_t$

B. $HD_s = HD_t$

III. AXIAL PUMPS

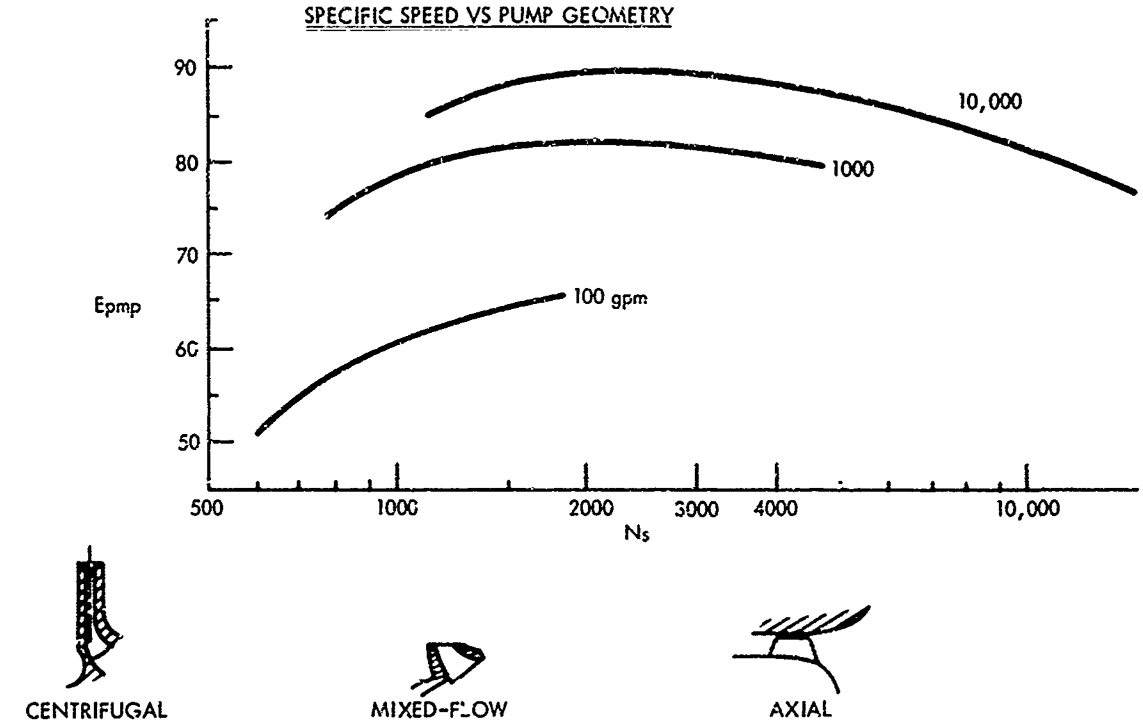


A. $Q_s = Q_1 = Q_2 = Q_t$

B. $HD_t = HD_1 + HD_2$

C. $HD_s = HD_t / Stgs$

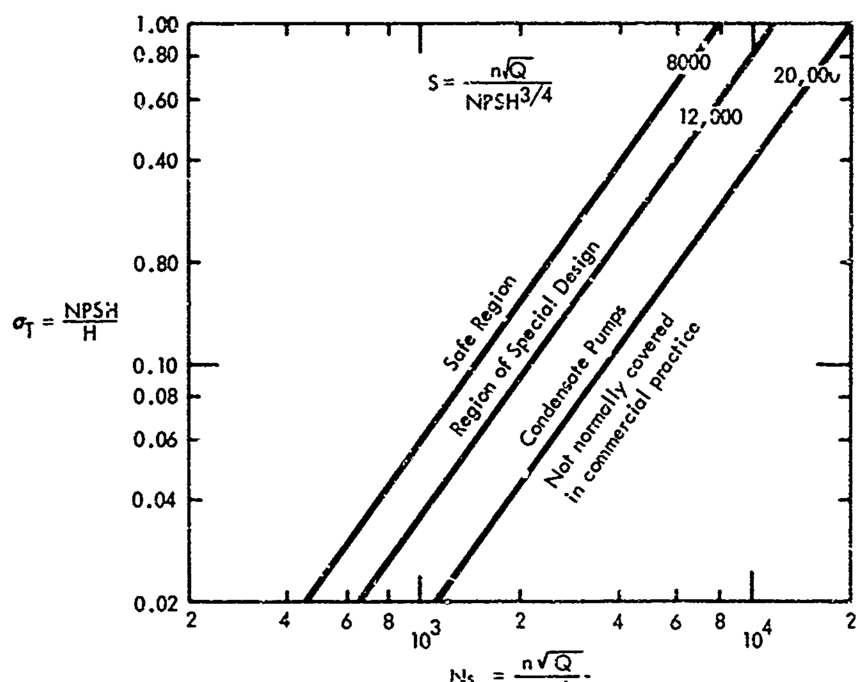
FIG. 12



N_s = RPM $\sqrt{gpm}/HD_s^{.75}$ (18)

RPM = $N_s\ HD_s^{.75}/\sqrt{gpm}$ (19)

FIG. 13



NPSH = HDav - Pvap (20)

σ = NPSH/HD_s (21)

SUCT = RPM $\sqrt{gpm}$/NPSH$^{.75}$ (22)

CAVITATION

FIG. 14

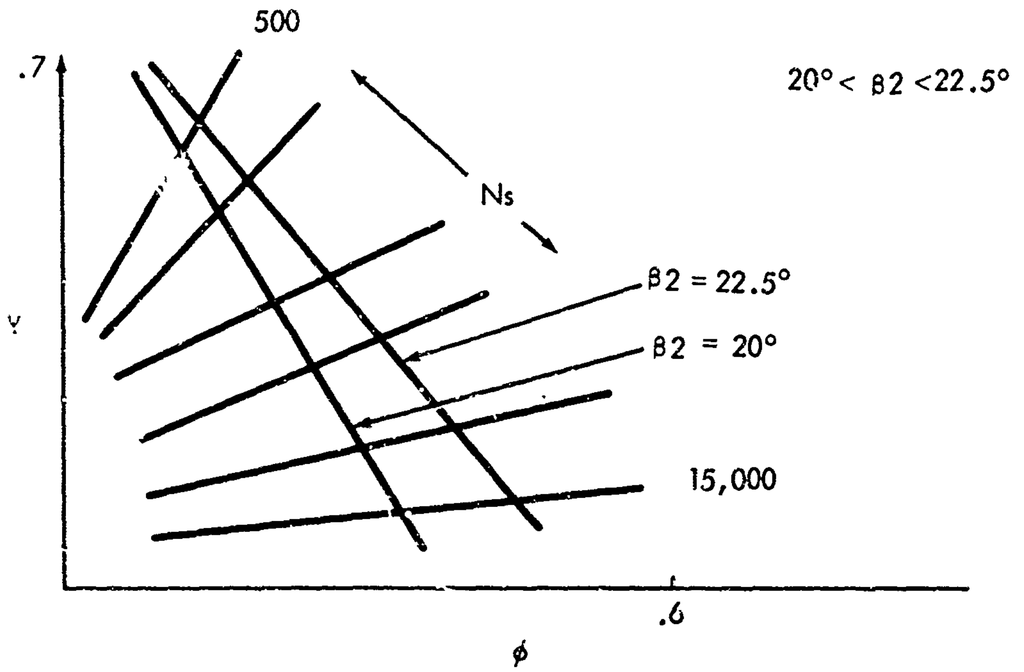
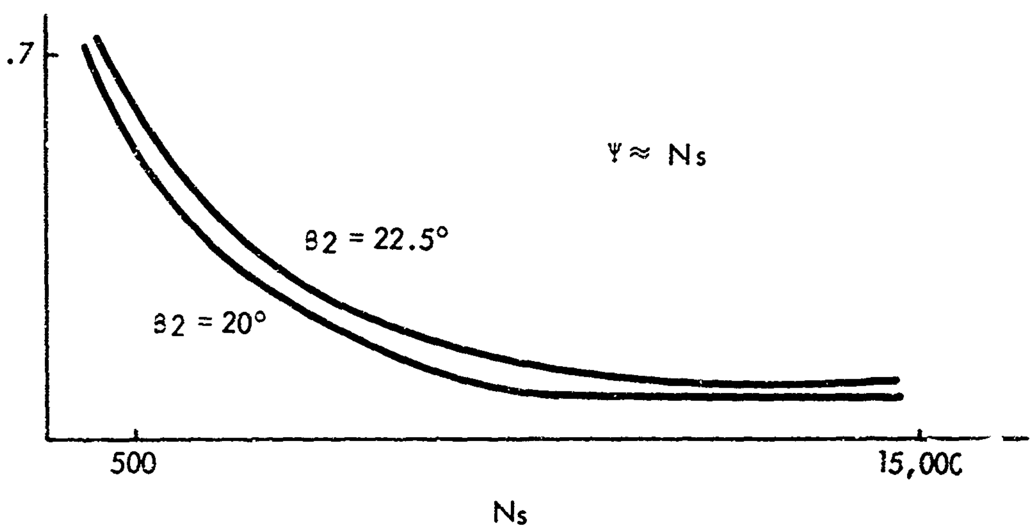


FIG. 15



$$\Psi = \frac{HD_s}{U2^2/g} \qquad (23)$$

$U2^2 = g\ HD_s/\Psi$, Ψ a function of Ns

$$Dimp = U2 \times 60/\ \pi \times RPM \qquad (24)$$

FIG. 16

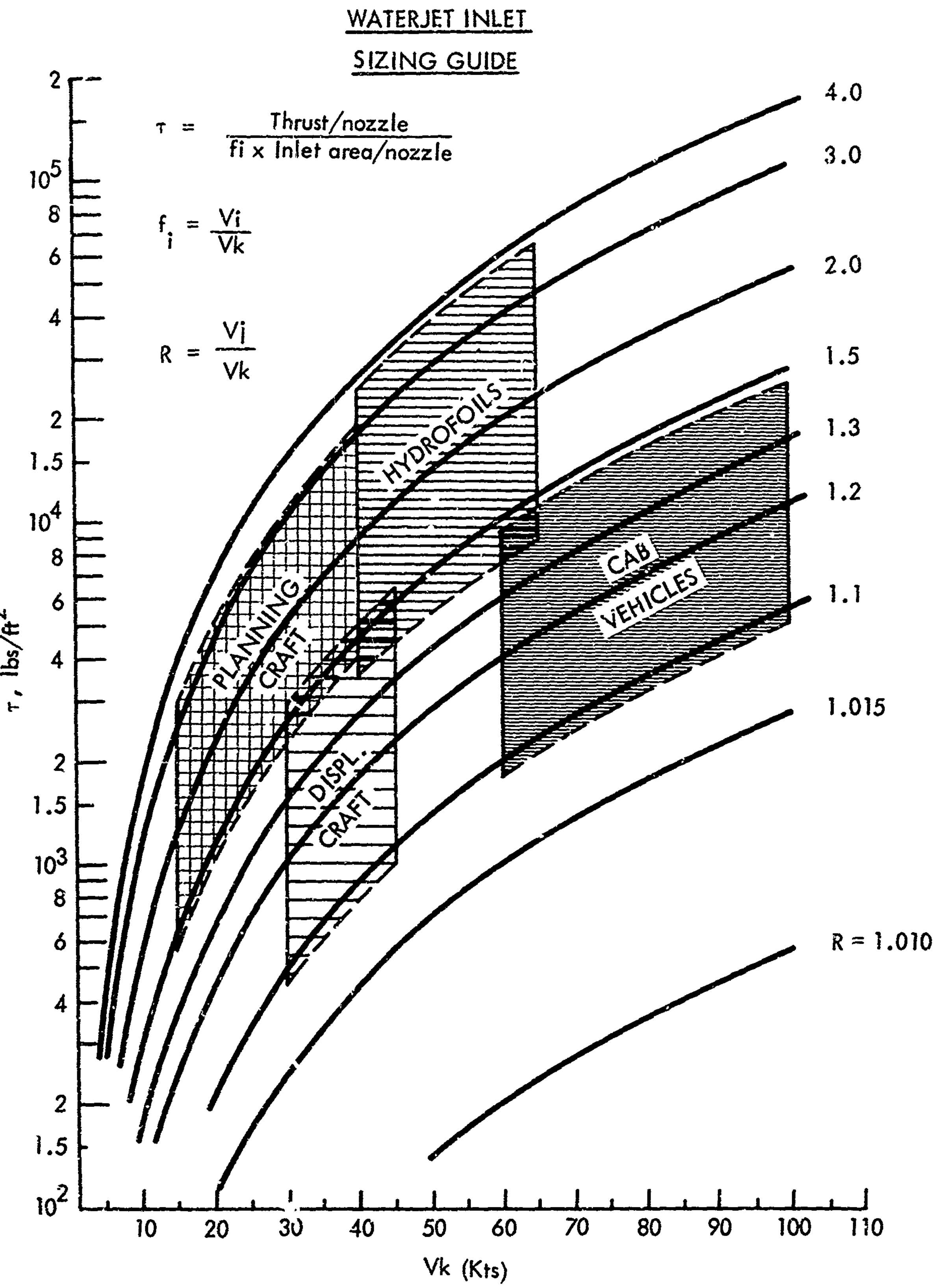

WATERJET INLET
SIZING GUIDE
τ = Thrust/nozzle / fi x Inlet area/nozzle
fi = Vi / Vk
R = Vj / Vk
HYDROFOILS
PLANNING CRAFT
DISPL. CRAFT
CAB VEHICLES
4.0
3.0
2.0
1.5
1.3
1.2
1.1
1.015
R = 1.010
τ, lbs/ft²
Vk (Kts)
10
20
30
40
50
60
70
80
90
100
110

FIG. B.1

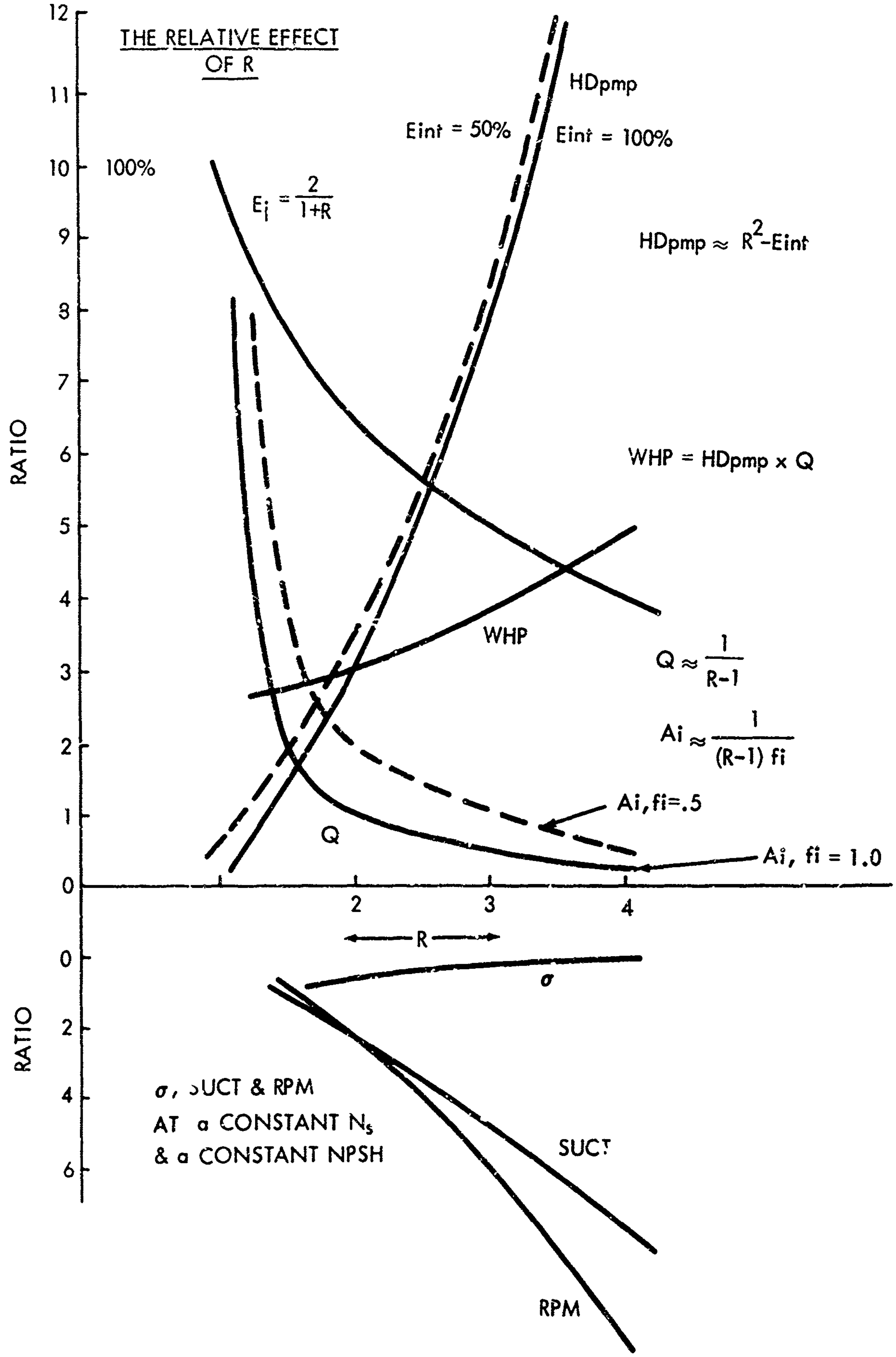

THE RELATIVE EFFECT OF R
12
11
10
9
8
7
6
5
4
3
2
1
0
RATIO
100%
$E_i = \frac{2}{1+R}$
Eint = 50%
Eint = 100%
HDpmp
$HDpmp \approx R^2 - Eint$
WHP = HDpmp x Q
WHP
$Q \approx \frac{1}{R-1}$
$Ai \approx \frac{1}{(R-1)\ fi}$
Ai, fi=.5
Ai, fi = 1.0
Q
2
3
4
R
0
2
4
6
RATIO
σ
σ, SUCT & RPM
AT a CONSTANT N_s
& a CONSTANT NPSH
SUCT
RPM

FIG. B.2